



# Intelligent Engine Systems

## Bearing System

*Anant P. Singh*  
*General Electric Aircraft Engines, Cincinnati, Ohio*

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# **Intelligent Engine Systems**

## **Bearing System**

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## **2.2 Work element: Bearing System**

### **2.2.1 Background, Technical Approach, and Proposed System**

#### **2.2.1.1 Background**

The failure of high-speed bearing continues to be one of the leading causes of In Flight Shut Downs (IFSD), and Unscheduled Engine Removals (UER). The failure of differential (or inter-shaft) roller bearings often results in class A mishaps for military engines. The failure of inter-shaft bearings is very difficult to detect with conventional lubrication system chip detection systems because of very difficult chip transportation. Traditional bearing health monitoring has utilized magnetic chip detectors for decades. More recently (last 10 years or so) quantitative debris monitors have been introduced which provide an on-line indication of debris size and rate of production. However, these devices have not shown the desired effectiveness for the inter-shaft bearings, where both inner and outer races rotate to create a field of centrifugal force making a tortuous path for the bearing debris.

Widely used vibration, and/or acoustic emission measurements at best can only diagnose the spell initiation for a very simplified non-differential roller or ball bearing. Identifying the bearing component frequencies among all the other excitations, and noise levels present in the real engine environment becomes increasingly difficult when the changing loads also vary the slip rates, and the cage speeds. This further complicates the basic “component damage” frequency relationship of any simplified vibration diagnostics.

The main objective of this program was to develop a reliable bearing failure detection system for the inter-shaft roller bearings. The previous phase of the current program identified, evaluated, and then successfully tested the selected rotating sensors for the purpose of differential bearing diagnostics. The diagnostic sensitivity with respect to different operating parameters was also tested. The key tasks, and major efforts planned for the current bearing diagnostic and health monitoring system development effort phase are the further continuation of the previous efforts.

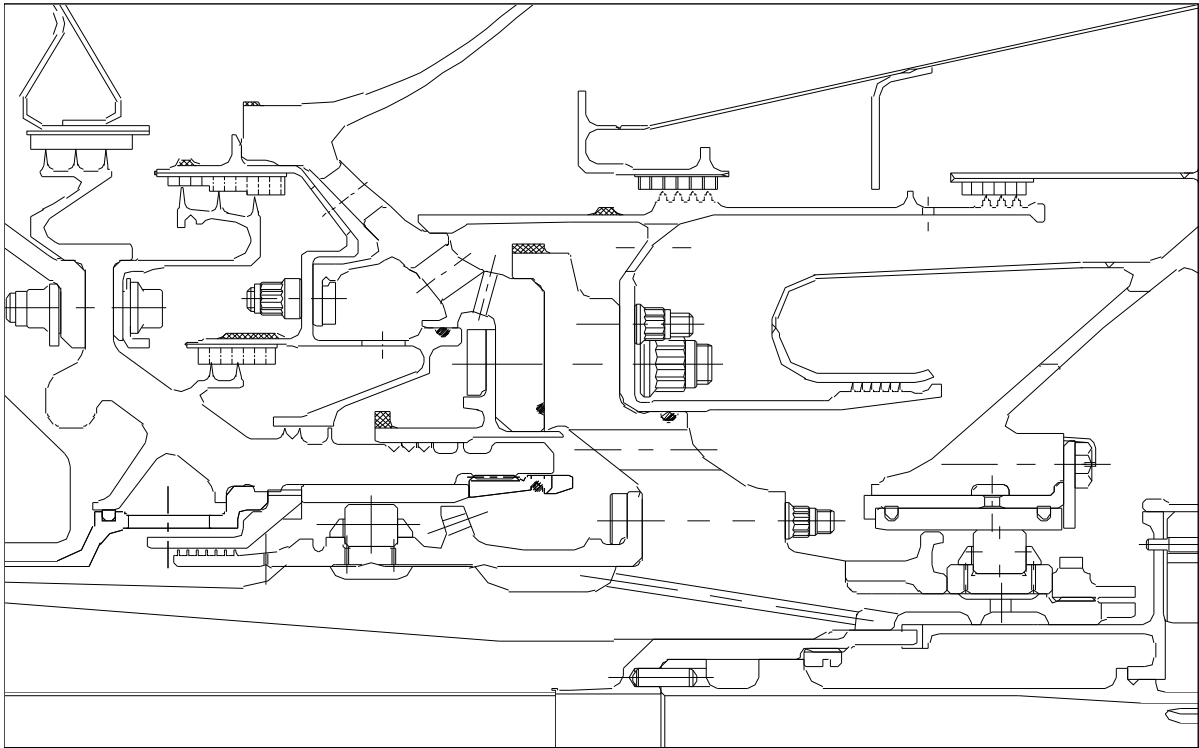
#### **2.2.1.2 Technical Approach**

The overall requirements necessary for sensing bearing distress and the related criteria to select a particular rotating sensor were established during the phase I. The current phase II efforts performed studies to evaluate the “Robustness and Durability Enhancement” of the rotating sensors, and to design, and develop the “Built-in Telemetry System” concepts for an aircraft engine differential sump.

A generic test vehicle that can test the proposed bearing diagnostic system was designed, developed, and built. The Timken Company, who also assisted with testing the GE concept of using rotating sensors for the differential bearing diagnostics during previous phase, was selected as a sub-contractor to assist GE for the design, and procurement of the test vehicle. A purchase order was prepared to define the different sub-tasks, and deliverables for this task. The University of Akron was selected to provide the necessary support for installing, and integrating the test vehicle with their newly designed test facility capable of simulating the operating environment for the planned testing. The planned testing with good and damaged bearings will be on hold pending further continuation of this effort during next phase.

### 2.2.1.3 Differential Bearing Diagnostics

Several different inter-shaft roller bearing configurations for the commercial as well as the military applications were reviewed. A typical differential sump for a commercial application is shown in Figure 2.2.1.3 -1. A variety of bearing fault detection techniques referred in the literature, and sometimes successfully used for the industrial bearing applications did not find much utilization with aircraft engine bearings.



**Figure 2.2.1.3-1: A typical differential sump**

The vibration diagnostics to identify a bearing fault becomes increasingly difficult under the aircraft engine core sump environment due to high speeds, presence of several frequencies (both synchronous and non-synchronous), circuit path/lead-out complexities, and harsh thermal surroundings. The challenge to measure direct vibration signals from the distressed bearing becomes overwhelming when the bearing happens to be a differential with both the races rotating co- or counter-clockwise, and the sump configuration does not allow the mounting of any non-contacting probe. Future non-differential core bearings, most likely will also have a damper with a similar challenge of not getting a proper signal transmission from the stationary race. Additionally, it is also considered vital to develop and acquire as much bearing fault diagnostic data and experience for the complex configurations, if reliable prognostic approaches are to be viable.

The need to explore and evaluate other innovative means to monitor bearing health appears to be a necessity. Other means may also include the measurement of lube supply and scavenge temperatures for a particular sump.



The key elements for the proposed diagnostics and prognostics system include the following:

- Rotating sensors, and installation
- Built-in telemetry system
- Test data based algorithms, and related control system

#### **2.2.1.4 Selection of Rotating Sensors for Bearing Diagnostics**

The suitable sensors that can be used for the differential bearings are very limited. The complexities of any vibration based diagnostic scheme, somewhat hostile sump operating environment, plus the fact that space is very limited and neither inner nor outer race is stationary, makes it very challenging. The key sensors for the proposed diagnostic scheme were identified, and successfully tested.

#### **2.2.2 Rotating Sensors Installation and Durability Enhancement**

The key challenge for rotating sensors always happens to be their ability to survive the operating conditions, and the harsh sump thermal environment. A review of the failure modes related to these sensors used for the sump applications was conducted. Several guidelines intended to enhance durability were prepared.

#### **2.2.3 Conceptual Designs for the “Built-in Telemetry System”**

The on line monitoring of any differential bearing cannot be accomplished without a robust telemetry design that can transmit the signals to stationary domain to use as a diagnostic as well as prognostic tool. The conventional slip ring configurations used during the engine development testing and related to the sump area are usually complex, and are also limited in its capability to withstand the harsh environment.

Therefore, the basic concepts for a telemetry system that can be built in the limited space associated with a differential sump, and can also withstand its thermal and operating environment, were developed. The basic design requirements for a built-in telemetry system for generic applications covering both the -co, and -counter rotations were established.

The feasibility of developing such concepts to accomplish a production version of the telemetry system that can meet the robustness, and life requirements was also checked by visiting and working with the telemetry vendors. The basic application definition for the required telemetry system was established.

The challenges identified for developing a built-in telemetry system for the differential sump included the following:

- **Survivability**
- **Development**
- **Bandwidth**
- **Circuitry**

## **2.2.4 Design and Build an Instrumented Differential Bearing Test Vehicle**

The overall program work scope for this sub-task consisted of several efforts with a key objective to design, build, and instrument the test vehicle. The proposed differential bearing test rig has been designed to be generic in nature to support either a counter or a co rotating inter-shaft bearing application. The co- and counter-rotating inter-shaft bearing operating conditions represented the typical GE co- and counter- rotating engine experience. The full-scale instrumented test vehicle was to be finally installed at the university of Akron facility to simulate the operating environments. Future phases of the propulsion 21 efforts will test the proposed diagnostic, and health-monitoring systems developed during current phase.

### **2.2.4.1 Establish Simulation Requirements**

The basic differential bearing, although very similar to the bearing used during the initial phase testing was expected to be capable of supporting the counter as well as the co rotating operations. The differential bearing power requirements for the counter rotating operation are expected to be higher than the co rotating operation. The differential bearing design will simulate the internal radial clearance IRC, and the resulting radial load requirements at Idle, cruise, and take-off conditions of a typical mission cycle. The HP shaft, driven by the HP drive system will be supported by another standard roller bearing and a ball bearing. The lube supply through the inner race, and the related scavenge arrangement are to simulate the typical sump condition at key mission segment points like idle, cruise, and take-off. The temperature limits for scavenge, and soak-back were established. The LP shaft supported by the differential bearing, another roller bearing, and a ball bearing to take an axial thrust could be driven by the LP drive system. The LP shaft will also be simulated to have sump wall separation from the support bearing area. The HP shaft having the inner race of the inter-shaft bearing could be supported by a roller, and a ball bearing will be driven by the HP drive. The key simulation goals to be accomplished are summarized in the following:

1. Bearing IRC optimized to best reflect Idle, cruise, and T/O conditions.
2. Bearing 1G and resultant radial loads.
3. Bearing speeds at idle, cruise, and T/O conditions
4. Bearing lube supply through inner race, and scavenge systems
5. Provisions to have the rotating sensors, instrumentation, and the possible lead out arrangements
6. Bearing fault detection process

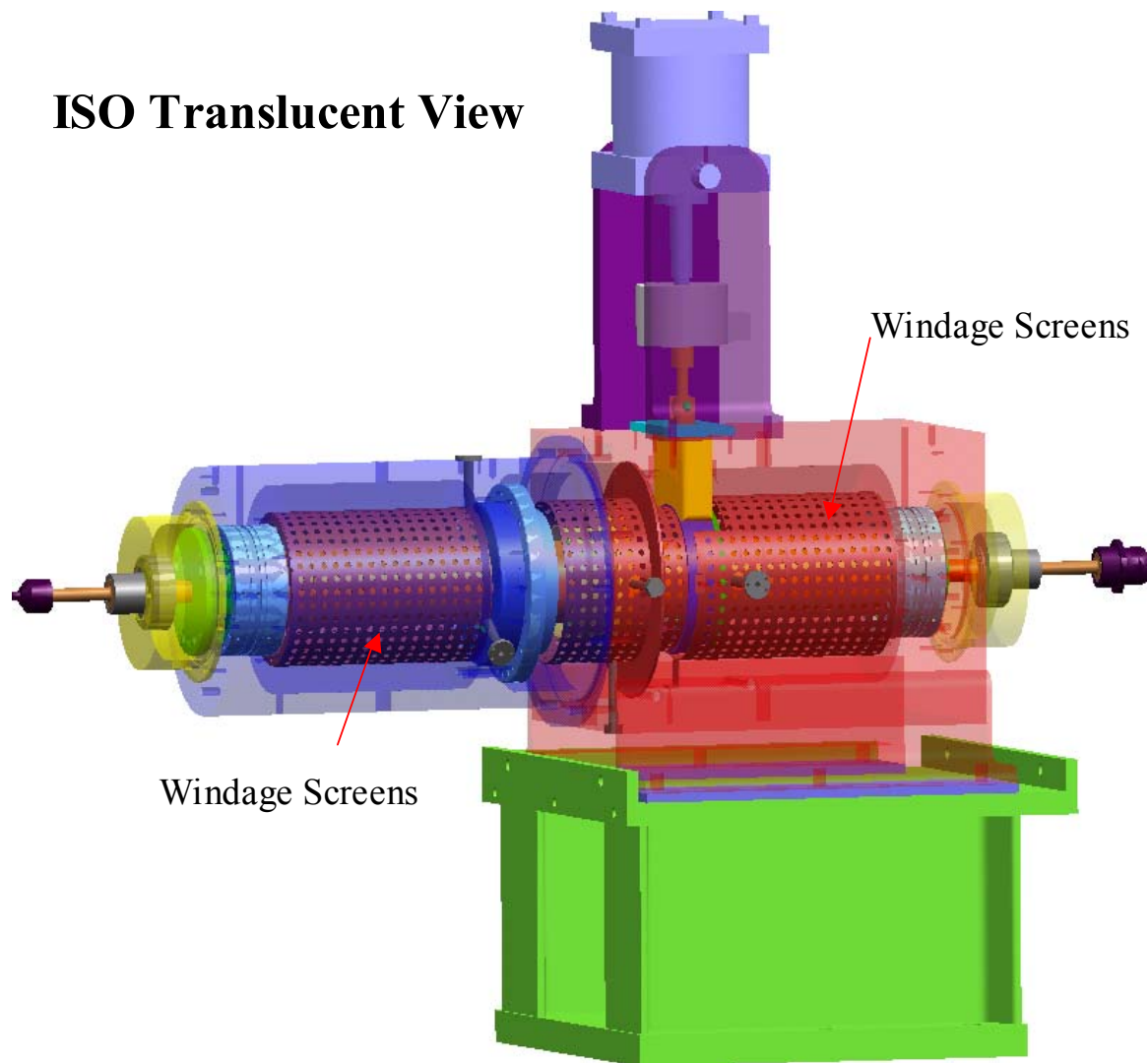
The key design requirements for the test vehicle capable of testing a variety of differential bearing applications were established.

The test vehicle capable of supporting both the co-, and counter- rotating operations simulated the test bearing IRC, and the resulting radial loads at idle, cruise, and take-off conditions of a typical mission cycle for three different applications. The requirement details were further modified to be compatible with the overall work-scope.

### **2.2.4.2 Finalized Layout of the Proposed Test Vehicle**

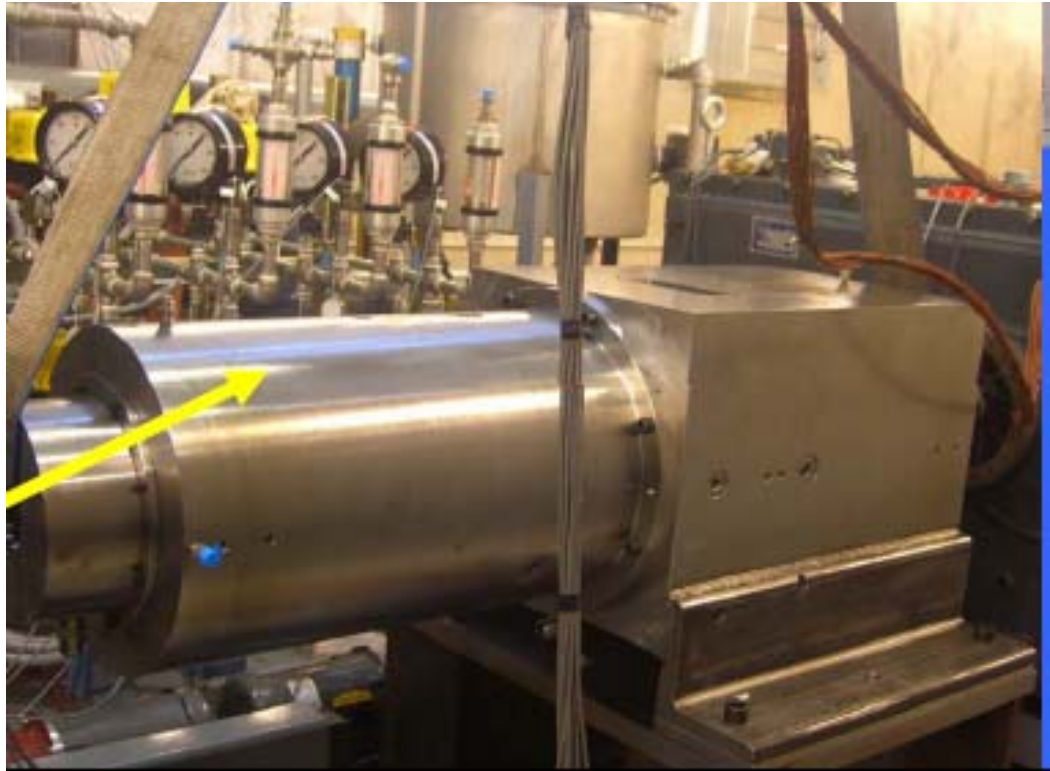
The ISO Translucent view of the test vehicle is shown in the following.

## ISO Translucent View



2.2.4.2-1: ISO Translucent view of the test vehicle

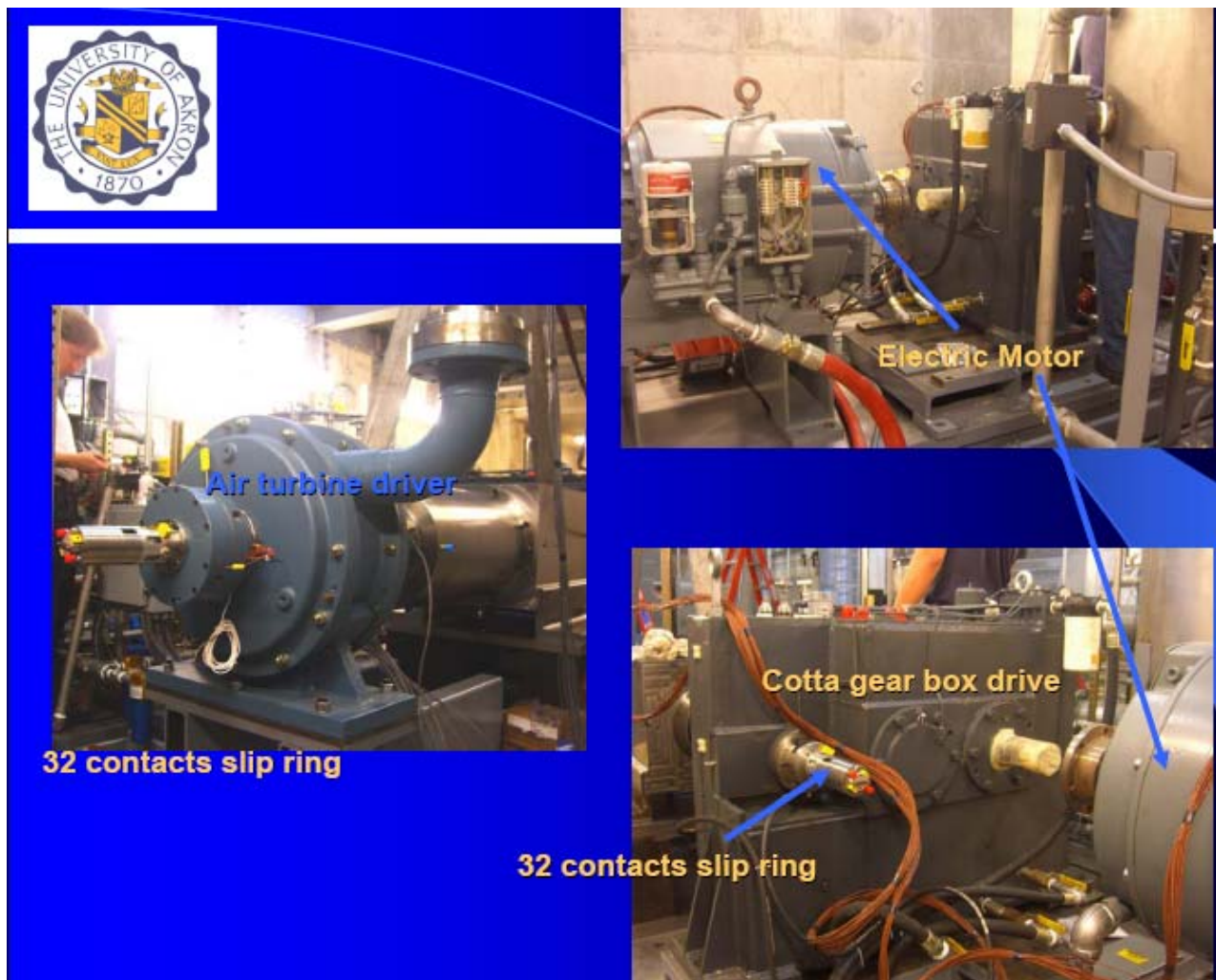
**Test Head  
Housing**



**2.2.4.2-2: Picture showing the test head housing, and lube system details**

### **2.2.4.3 Test Vehicle Instrumentation**

The instrumentation, lead-outs, and slip ring telemetry details for the test vehicle were finalized after several consultation meetings with different teams. The detailed design review for the finalized test vehicle was successfully conducted. The action items from this review were completed, and recommendations to ease the assembly, and the instrumentation plan were implemented.



**Figure 2.2.4.3-1: Slip Ring arrangements at both ends**

#### **2.2.4.4 Lube System Design for the Test Vehicle**

The lube system design for any differential sump usually is more challenging than the conventional sumps because of the centrifugal field. The lubricant oil for the test vehicle was delivered at several inlet points, and then supplied to the bearings. The duplex ball bearing received oil through a grooved spacer between the outer rings. The oil was directed through axial holes in the spacers and underneath the outer raceway of both rows in each set. Radial oil pins delivered lubrication to the cylindrical slave bearings. The orifice diameters were sized to provide the required flow rate to maintain a desired operating temperature. The oil stream was directed between the cage flange and the inner rings. The lubricant used was Mobil Jet II oil that met the MIL-L-23699.

The analytical predictions for the heat generation covering the worst operating, and loading conditions for each bearing were used to quantify the required oil flow rates. The flow rates were established for a temperature rise target of 40° F. The counter rotation operation as compared to the -co rotating operation did require more flow to balance the increased heat generation. The allowable max flow rate was predicted for the test bearing when to support a max load, max speed operation with a counter clockwise rotation.

#### **2.2.4.5 The Lube Distribution System for the Test Bearing**

The lube delivery and distribution scheme for supporting the test bearing was established. The test bearing located in the middle of the test head, was provided lubricant through two nozzles located in the oil jet plate mounted at the forward side of the HP shaft. The jets consisted of 7/64" drilled holes spaced 180 degree apart and protrude inside the face of the rotating ball bearing clamp. The oil stream is targeted at the chamber located just under the duplex bearings. The oil impinges on the rotating shaft, and is thrown outward by the centrifugal force to form an annulus of oil around the ID surface of the capture section. Locating the oil jet stream inside the rotating face helps ensure that rotating shaft windage does not impede the oil stream from hitting the target. The ball bearing clamp on the HP shaft has a shoulder that is angled and serves to keep the oil in capture section. This helps the oil to progress through eight equally spaced 1/4" holes that are drilled through the web section tangent to the HP shaft ID. This allows the lubricant oil to flow into the ID of the HP shaft while being separated from the slip ring lead-wires by the slip tube. A computational fluid dynamics (CFD) analysis was performed to ensure if the axial holes through the shaft web could pass all the oil delivered by the jets into the shaft bore. The oil arriving at the inboard end of the HP shaft flows into a circumferential trough located in the HP shaft ID just before the test bearing. A series of equally spaced eight axial grooves were machined in the HP shaft bore, and had one 3/32" radial drilled hole per slot to transfer the oil outward toward two circumferential grooves in the test bearing bore. The cylindrical support roller bearings are lubricated through 1/16" drilled holes in an oil pin.

The installed sensors on the test bearing were protected from the oil paths of the other bearings by having close clearance sheet metal Windage shields on either side. The basic purpose of the Windage screen is to dissipate the energy of the centrifuged oil off the HP and LP shafts during operation. Otherwise, the oil could induce vibration and eventually lead to a cavitation inside the test head housing. On the LP shaft the shield is mounted to an oil Windage sheet that surrounds the outer race of the test bearing. On the HP shaft the shield is mounted to the Windage screen attached to the HP cylindrical bearing clamp.

### **2.2.4 The Test Plan**

The detailed test plan covering the objectives, simulated mission definition, types of induced defects, and the sensitivity with respect to fixed and variable test-parameters was established.

#### **2.2.5.1 Test Objectives**

The overall test objectives for the **initial break-in / Mechanical check out test** included the following:

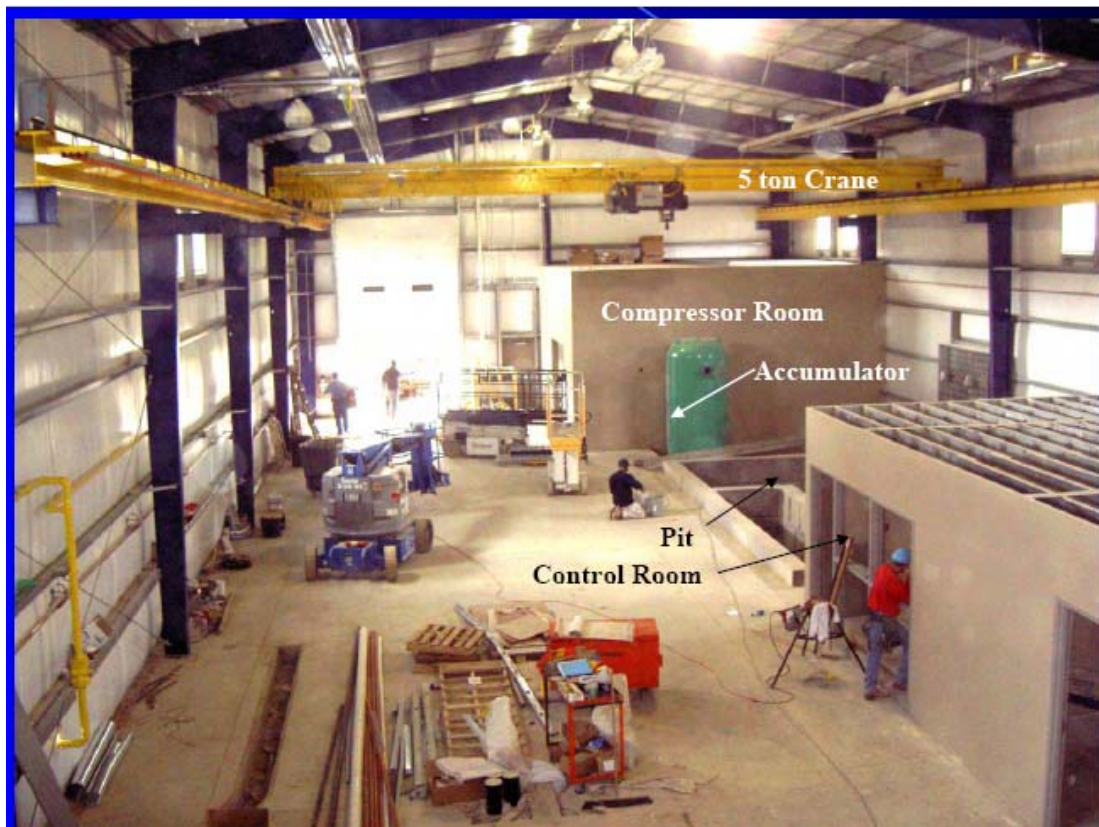
1. Check if the instrumentation is working as expected under the simulated environment of different applications.
2. Check if the arrangements / mechanisms to vary the radial loads, speed, oil supply flow rate are functional, and meet the design / test intent.



3. Check, and ensure the proper functioning of the slip ring arrangements for the LP, and HP rotors.
  4. Check if all other facility hardware / equipments are working properly.
- The objectives for the planned testing related to a “Differential Bearing Diagnostic Performance” for three different applications included the following:
1. Check test bearing diagnostics performance at simulated operating environment at three mission points namely idle/taxi, cruise, and take off for the nominal static radial load
    - Steady state points at prescribed LP, and HP speeds simulating the idle, cruise, and T/O conditions during decal
    - Steady state points at prescribed LP, and HP speeds simulating the idle, cruise, and T/O conditions during acceleration
  2. Impact of static radial load variation on the diagnostic sensitivity at each mission points namely idle, cruise, and T/O
    - Vary the resulting radial load by +/- 50% from the prescribed nominal value
  3. Impact of varying the oil flow rate and/or any other parameter

## 2.2.6 Test Vehicle Integration With the Akron University Test Facility

A new test facility, compatible for running the newly designed, test vehicle for the desired operating, and loading conditions was built. This effort supported by the Akron University was part of the Propulsion 21 program. Figure 2.2.6-1 shows the newly built test facility at University of Akron.



**Figure 2.2.6-1: Akron University Test Facility Building in Final Stages**

### 2.2.6.1 System Overview and Different Sub-systems

The test facility that can support the planned testing, and could provide support for any future extension of current developments included several sub-systems to be developed, and established. There key areas included the drive system, lube system, and the test & data acquisition system.

### 2.2.6.2 Drive Systems

The generic differential bearing applications required drive systems for both the LP and HP rotors with possibility to accomplish both -co, and -counter rotating operation. The HP drive with maximum speed of 20,000 rpm was driven by Barbour-Stockwell air turbine, and had only one direction of rotation. The characteristics for this radial inflow turbine included 60K rpm, and 25.6 FT-LBF torque. A large compressor and accumulator were installed to provide the airflow.

The LP drive system utilized the 150 HP electric motor with a dynamic eddy current clutch. This unit is capable of 50 to 1700 rpm range over a 540-to 5400 in-lb torque ranges. The drive coupled to a modified Cotta Transmission step-up gearbox provides both -co, and -counter rotating operation. The gearbox with a 1:13.1 ratio is capable of 655 to 22,270 rpm +/- rpm over a torque demand of 41.2 to 412 in-lbf. The Cotta increaser has a three parallel shaft / precision helical gear design incorporating anti-friction ball bearings and an integral lube pump. The pump, filtration and oil cooling is done with a closed loop system outside of the box and mounted to the base plate. The output shaft is hollow and the wires from the sensors mounted on the test bearing outer race run through the shaft to the high- speed slip ring that is mounted directly on the outboard side of the gearbox.

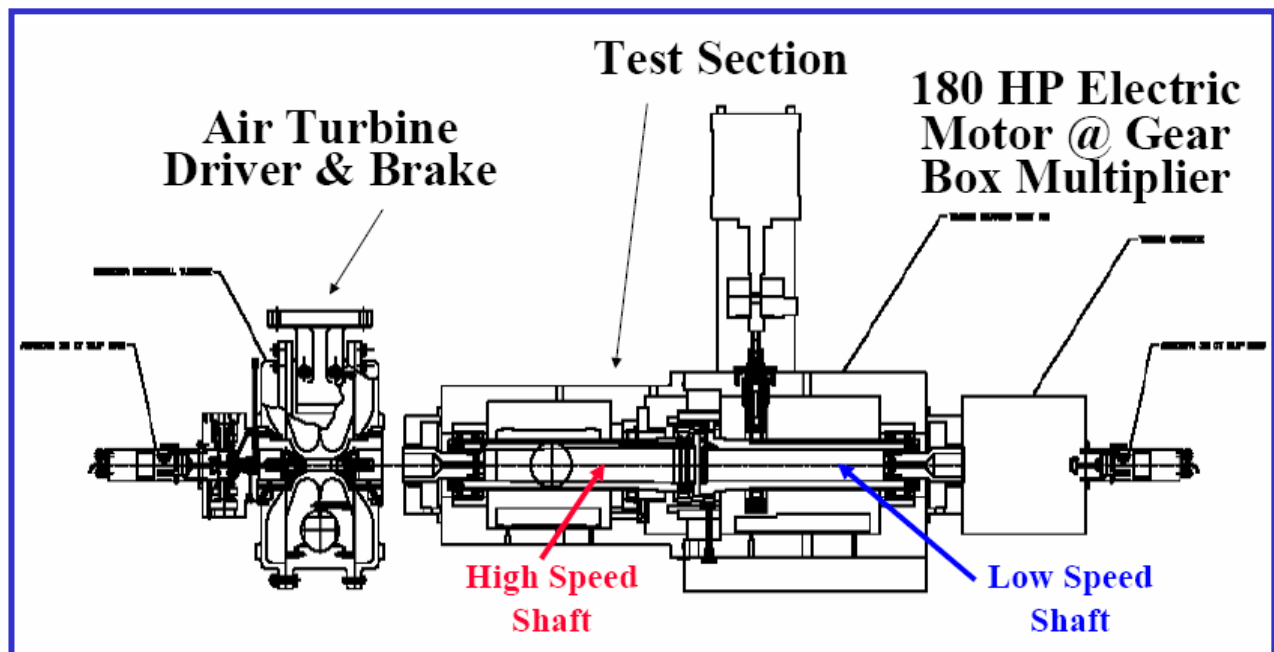


Figure 2.2.6.2-1: Schematic for the drive systems at both ends



### 2.2.6.3 Lube Systems

The lubrication system to supply Mobil Jet Oil II (MIL-L-23699) for the test vehicle was designed by the Akron University. The lube system scheme consisted of two main sections with separate oil reservoirs, pumps, oil heaters and return loops. The first section as shown in figure 2.2.6.3-1 supplied oil to HP and LP support bearings. A fifty-gallon tank used as an oil reservoir was installed with a heater to meet the lube supply temperature requirements. Temperature, pressures, and flow rate are to be monitored by gages located on each manifold outlet.

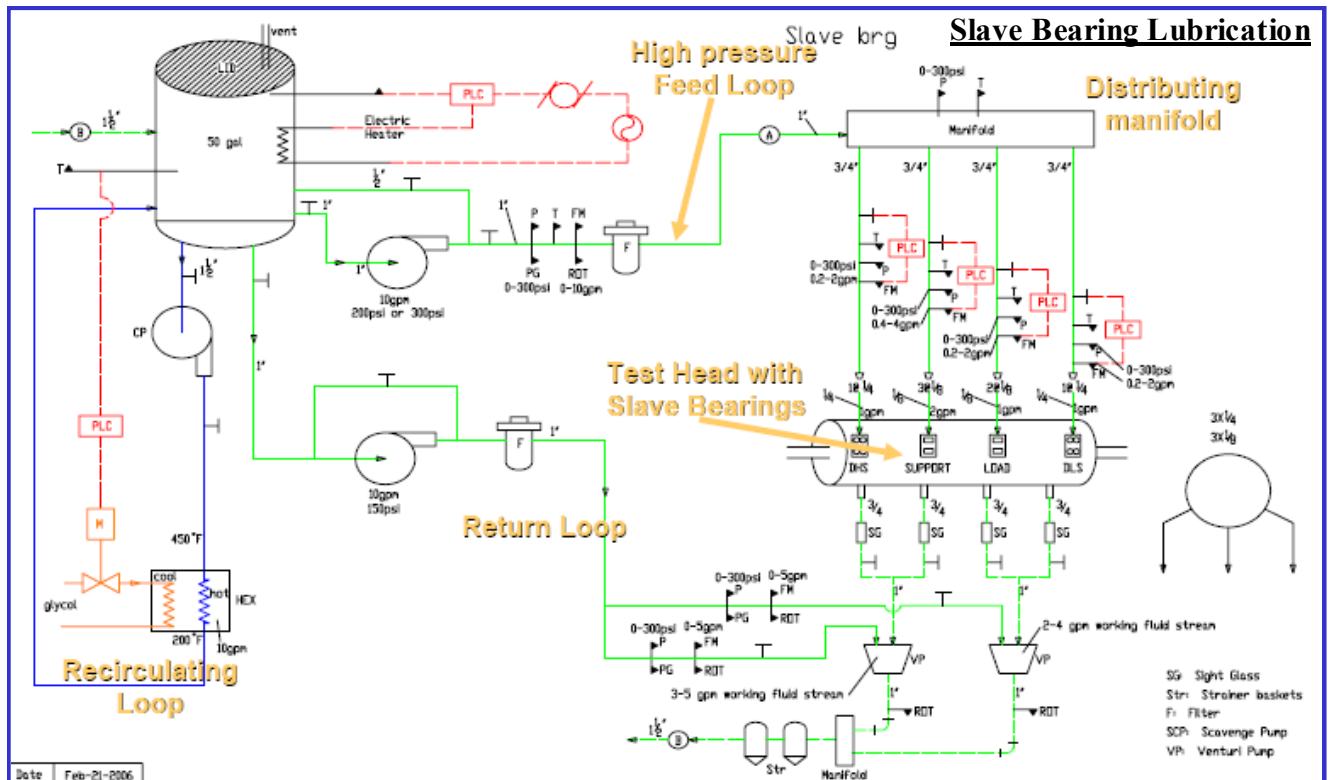


Figure 2.2.6.3-1: Schematic for the lube system supporting slave bearings

The second section of this system supplies lubricant for the test bearing as shown in figure 2.2.6.3-2. Oil flows into a separate manifold that distributes the oil to oil jet at the HP end of the test vehicle. The flow, pressure, and temperature data is to be monitored, and the flow rate could be adjusted as needed.

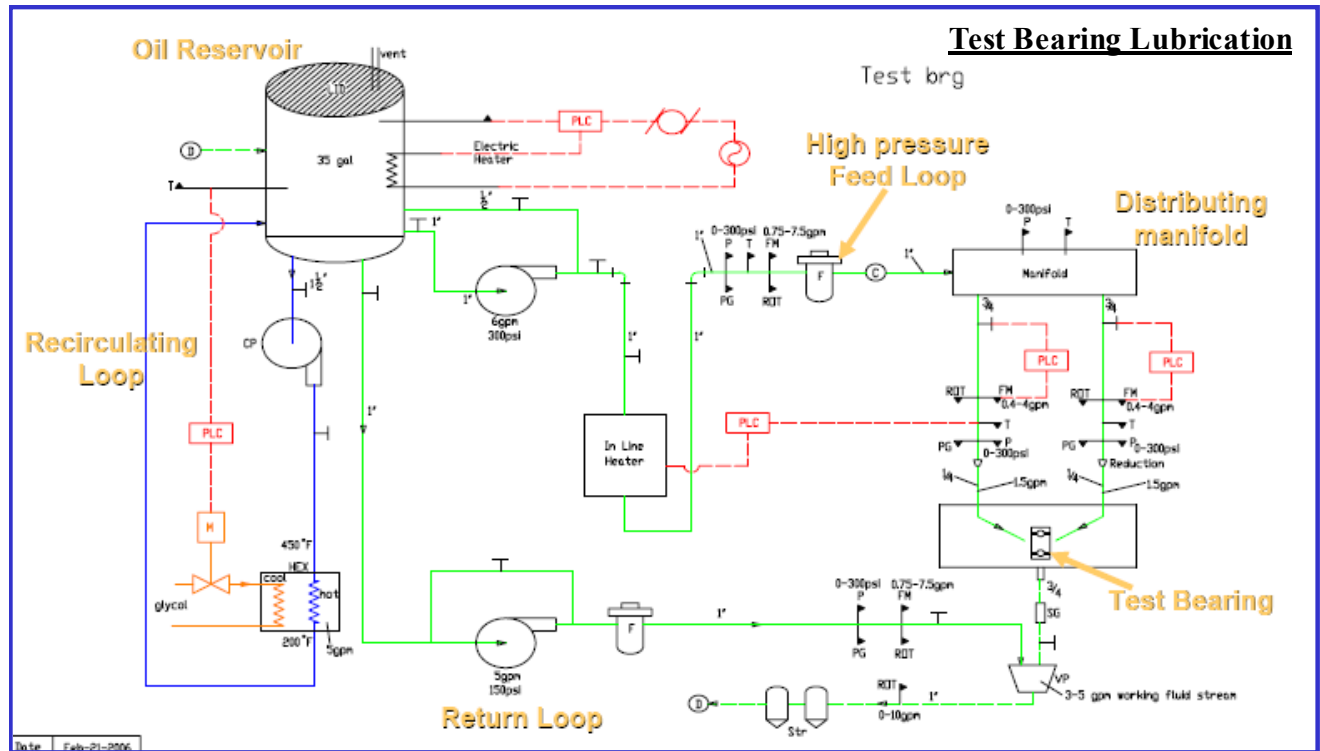


Figure 2.2.6.3-2: The lube-system schematic supporting test bearing



**Figure 2.2.6.3-3: Pictures showing the lube system details**

#### **2.2.6.4 Test Data Acquisition System**

Test data acquisition system was developed by Akron University for the new test facility. The low-speed data acquisition system as shown in figure 2.2.6.4-1 had NEFE 470 system with 256 channels. The University of Akron has completed the 100% implementation of 10 KHz, 256 channels. The high-speed data acquisition system had Labview, National instruments with 64 channels. The implementation of the 500 KHz/channel, 64channel data acquisition has been completed.

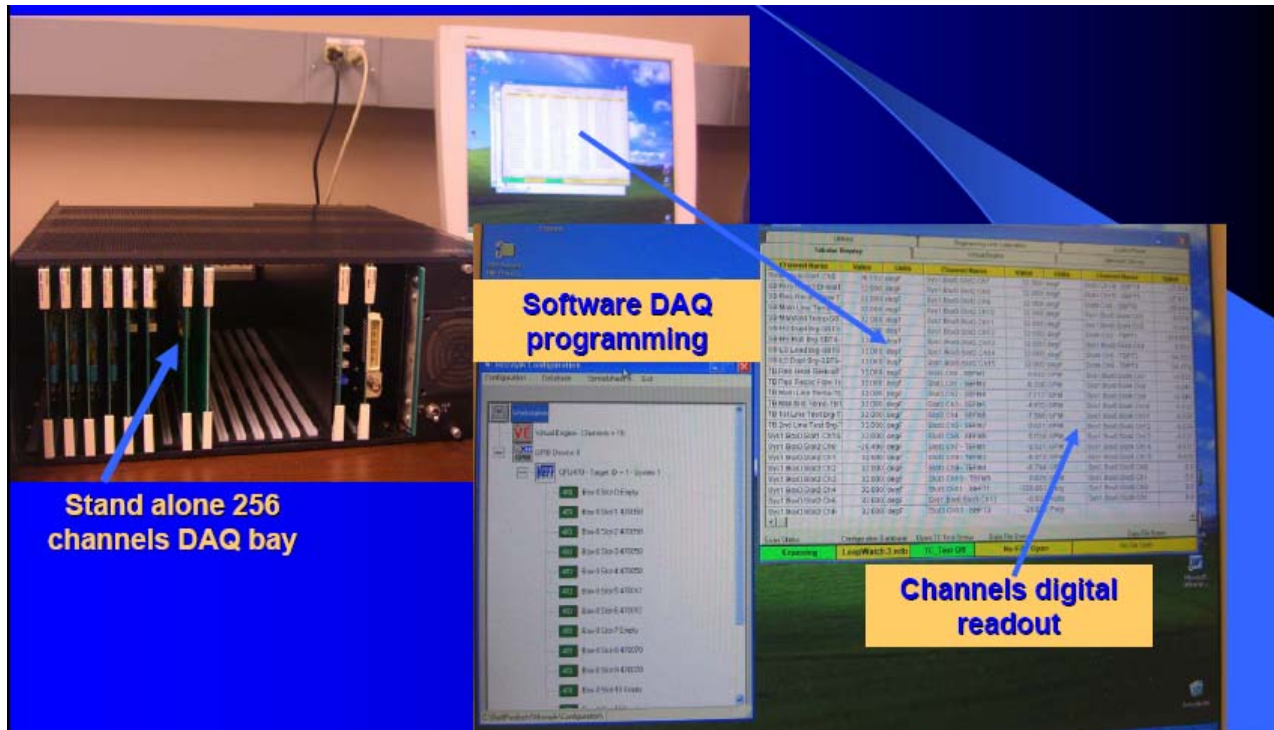


Figure 2.2.6.4-1: Pictures showing the data acquisition details

## 2.2.7 Recommendations for Future Efforts

- Complete the planned test using the new, and the defect induced test bearings to study and evaluate the following:
  - Sensitivity of different sensors, its location and orientation with respect to the type of defect, its location, and size for a particular bearing race
  - Damage detection sensitivity with respect to the resultant radial load, and the operating speeds
  - Differences with respect the -co, and the -counter rotating differential bearing operations
- Modify the present test rig to simulate an actual engine sump environment with pressurized labyrinth or carbon seals, and study the detection sensitivity with respect to different operating parameters
  - Test the sensor robustness under simulated sump thermal, loading, and operating environment
- Develop, procure, and test the “Sump Built-in Telemetry System” for different applications.
- Test the new differential bearing diagnostic and built-in telemetry systems on a real engine.

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| 14. ABSTRACT<br>The overall requirements necessary for sensing bearing distress and the related criteria to select a particular rotating sensor were established during the phase I. The current phase II efforts performed studies to evaluate the "Robustness and Durability Enhancement" of the rotating sensors, and to design, and develop the "Built-in Telemetry System" concepts for an aircraft engine differential sump. A generic test vehicle that can test the proposed bearing diagnostic system was designed, developed, and built. The Timken Company, who also assisted with testing the GE concept of using rotating sensors for the differential bearing diagnostics during previous phase, was selected as a sub-contractor to assist GE for the design, and procurement of the test vehicle. A purchase order was prepared to define the different sub-tasks, and deliverables for this task. The University of Akron was selected to provide the necessary support for installing, and integrating the test vehicle with their newly designed test facility capable of simulating the operating environment for the planned testing. The planned testing with good and damaged bearings will be on hold pending further continuation of this effort during next phase. |                  |   |                               |   |   |
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